AN INVESTIGATION OF SOME OF THE CHARACTERISTICS OF A JERK PUMP INJECTION SYSTEM FOR DIESEL ENGINES

ROBERT LEE MOORE

An Investigation of Some of the Characteristics
of a Jerk Pump Injection System for Diesel Engines

By

Robert Lee Moore, Jr.
B.S. (United States Naval Academy) 1930

Thesis

Submitted in partial satisfaction of the requirements for the degree of

MASTER OF SCIENCE

in

Mechanical Engineering

in the

Graduate Division

of the

University of California

Approved:

..........C. J. VOGT..........................

..........L. M. K. BOELTER..................

..........B. F. RABER......................
Committee in Charge

Deposited in the University Library ....
Date Librarian

June 1939

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS

STATEMENT OF PROBLEM

The purpose of this investigation was to determine the influence of pump speed, nozzle opening pressure, rack setting, and length of line on the discharge rate of the injection valve in a jerk pump injection system for Diesel engines. Since some of the above variables may be controlled by an operator for any given system, it is felt that any correlation, mathematical or experimental, between these several variables would be of value to the designer of the injection system as well as to the Diesel operator.

In addition to obtaining a correlation between the above variables, FOR A PARTICULAR INSTALLATION, it appeared desirable to determine whether or not these variables might be changed with respect to one another to obtain performance which might be predicted, and which would result in better operation.

With these objectives in mind the problem was attacked from a purely experimental point of view and the results are recorded herein.

[illegible]

[illegible] some of the above variables may be controlled by an operator for any given system, it is felt that any correlation, mathematical or experimental, between these several variables would be of value to the designer of the layout or system as well as to the Diesel operator.

In addition to obtaining a correlation between the above variables, FOR A PARTICULAR INSTALLATION, it appeared desirable to determine whether or not these variables might be changed with respect to one another to obtain performance which might be predicted, and which would result in better operation.

With these objectives in mind the problem was attacked from a purely experimental point of view and the results are recorded herein.

APPARATUS AND TEST PROCEDURE

A general view of the apparatus is shown in Figure 1, and a schematic diagram of the same apparatus is shown in Figure 4. In the above two figures the same letters are used to designate the same parts. Referring to either Figure 1 or Figure 4:

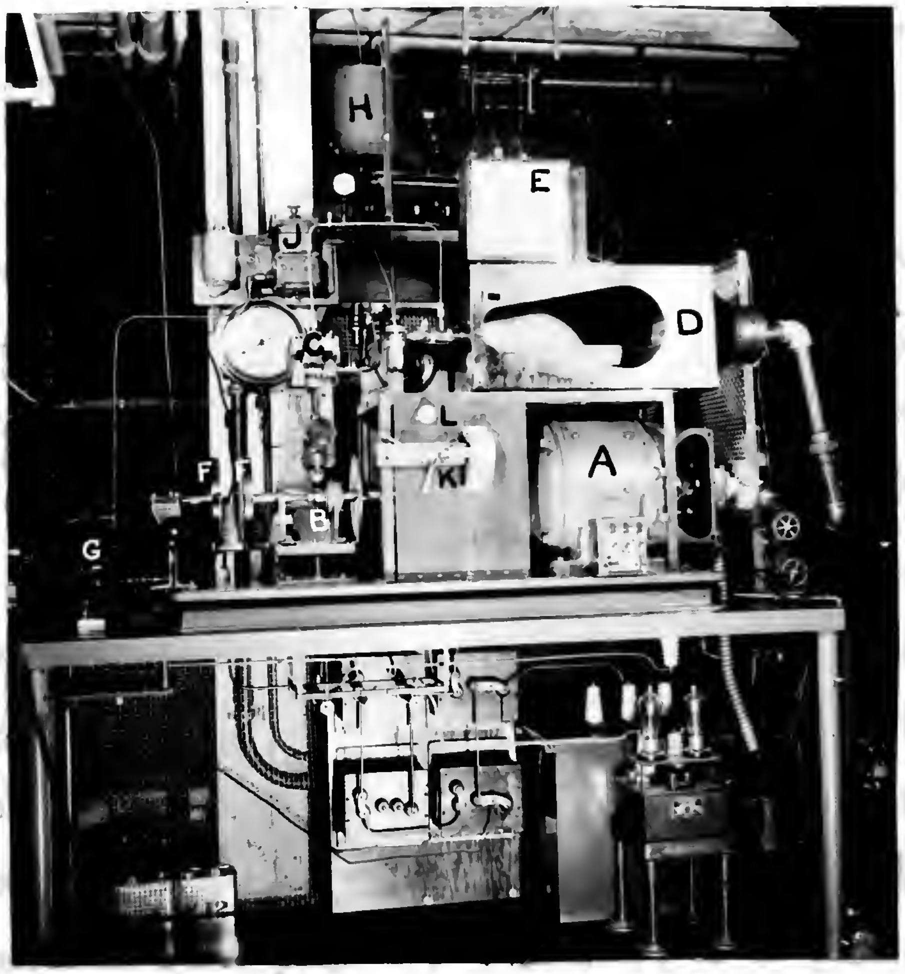


FIGURE 1.

"A" is a shunt motor having rheostatic speed control which drives the shaft to which is secured the cam "B", this cam in turn operates the plunger of the jerk pump "C". From the pump "C" the oil is discharged through a pipe line to the nozzle valve which is located in the box "D". A graduated scale is secured in box "D" as shown, by means of which the spray penetration may be determined. Mounted on

[illegible]

[illegible] of the apparatus is shown in Figure 1, and a [illegible] of the same apparatus is shown in Figure 2. In the above two figures the same letters are used to designate the same parts. Referring to either Figure 1 or Figure 2, [illegible]

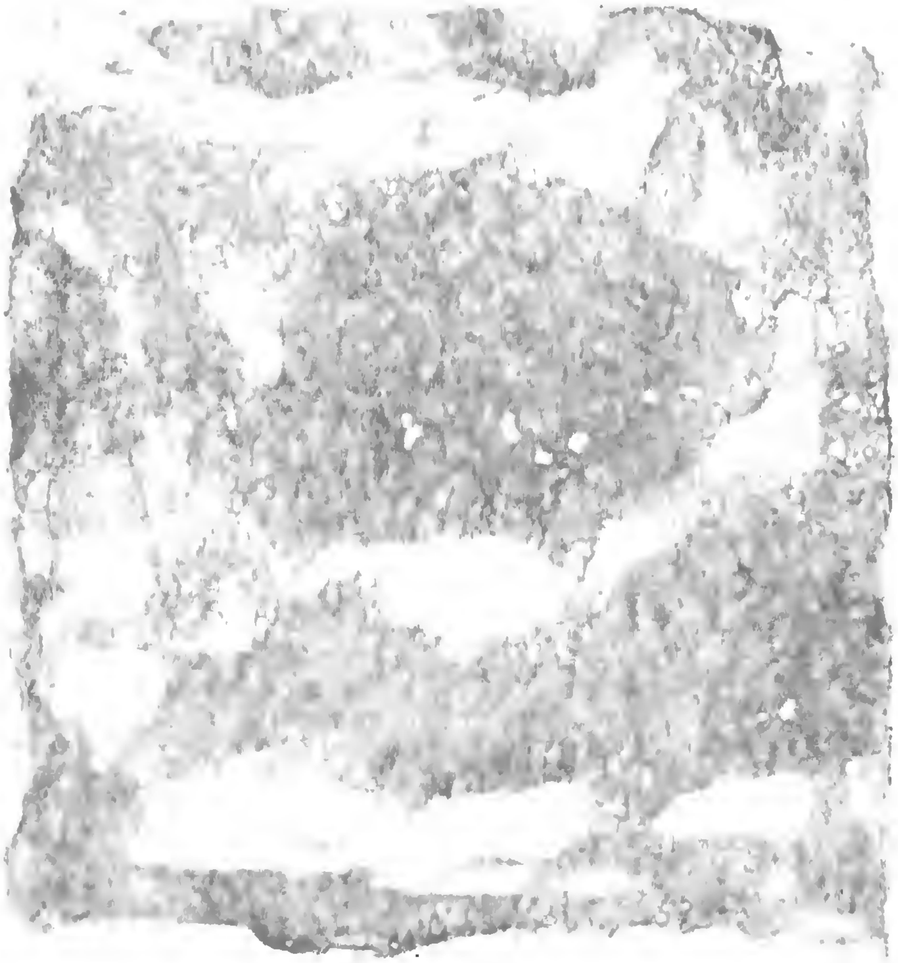


FIGURE 1.

"A" is a shunt motor having rheostatic speed control which drives the shaft to which is secured the cam "B", this cam in turn operates the plunger of the jerk pump "C". From the pump "C" the oil is discharged through a pipe line to the nozzle valve which is located in the box "D". A graduated scale is secured in box "D" as shown, by means of which the spray penetration may be determined. Mounted on

top of the box "D" is a stroboscopic neon light "E" which is made to illuminate the interior of the box "D" through a glass window in the top of "D". The time of the flashing of the neon light "E" may be made to occur at any desired angular shaft position by means of the graduated and variable rotary spark gap "F", one part of which is secured to the end of the motor shaft, while the variable part is mounted on the frame. A wiring diagram, Figure 5, shows the electrical connections between the spark gap "F" and the neon light "E". "H" is the oil reservoir located on one platform of a balance scale. Balance is obtained by placing weights of various magnitudes on the other platform; the instant of perfect balance being indicated by the flashing of a small neon light "L", Figure 1. This neon light is controlled by an electrical circuit through two small wires secured to either platform; the wires moving in and out of mercury baths, located under the platforms, as the balance changes position. From "H" the oil is delivered to the fuel pump "C" through the oil filter "J". It is to be noted that oil may be delivered to the pump "C" by gravity or under pressure by means of the variable speed pump "G". A pressure of 12 pounds per square inch was maintained at the suction side of the pump "C" throughout the investigation.

The pump used was a 10-millimeter "jerk-pump", having the conventional plunger scroll control. A cut-away picture of this type of pump is shown in Figure 2(B). The quantity of oil delivered by the pump may be varied by rotating the plunger "P" by means of the gear "G" which is secured to the plunger "P". Gear "G" is in turn

light "L". "R" is the oil reservoir located on one platform of a balance scale. Balance is obtained by placing weights of various magnitudes on the other platform; the instant of perfect balance being indicated by the flashing of a small neon light "L", Figure 1. This neon light is controlled by an electrical circuit through two small wires secured to either platform; the wires moving in and out of mercury baths, located under the platforms, as the balance changes position. From "R" the oil is delivered to the fuel pump "C" through the oil filter "F". It is to be noted that oil may be delivered to the pump "C" by gravity or under pressure by means of the variable speed pump "Q". A pressure of 12 pounds per square inch was maintained at the suction side of the pump "C" throughout the investigation.

The pump used was a 10-millimeter "jerk-pump", having the conventional plunger scroll control. A cut-away picture of this type of pump is shown in Figure 2(b). The quantity of oil delivered by the pump may be varied by rotating the plunger "P" by means of the gear "G" which is secured to the plunger "P". Gear "G" is in turn

actuated by means of an engaging rack, the position of which is controlled by the micrometer head "R", Figure 4. The end of the plunger "P" is actuated by the cam "B", Figure 4, by means of suitable linkage. The operation of the fuel pump "B", Figure 2, is as follows: as the plunger "P" moves from right to left, the space "E" becomes isolated when the flat plunger face reaches the left edge of the inlet channel "J". Further motion of "P" to the left causes the plunger to force oil from "E" past the check valve "V" until the left edge of the scroll space "S" reaches the right edge of the inlet channel "J". At this point the space "E" communicates directly with the inlet channel "J" through a small hole drilled from the plunger face to the scroll space "S", and the oil in space "E" is by-passed back to "J".

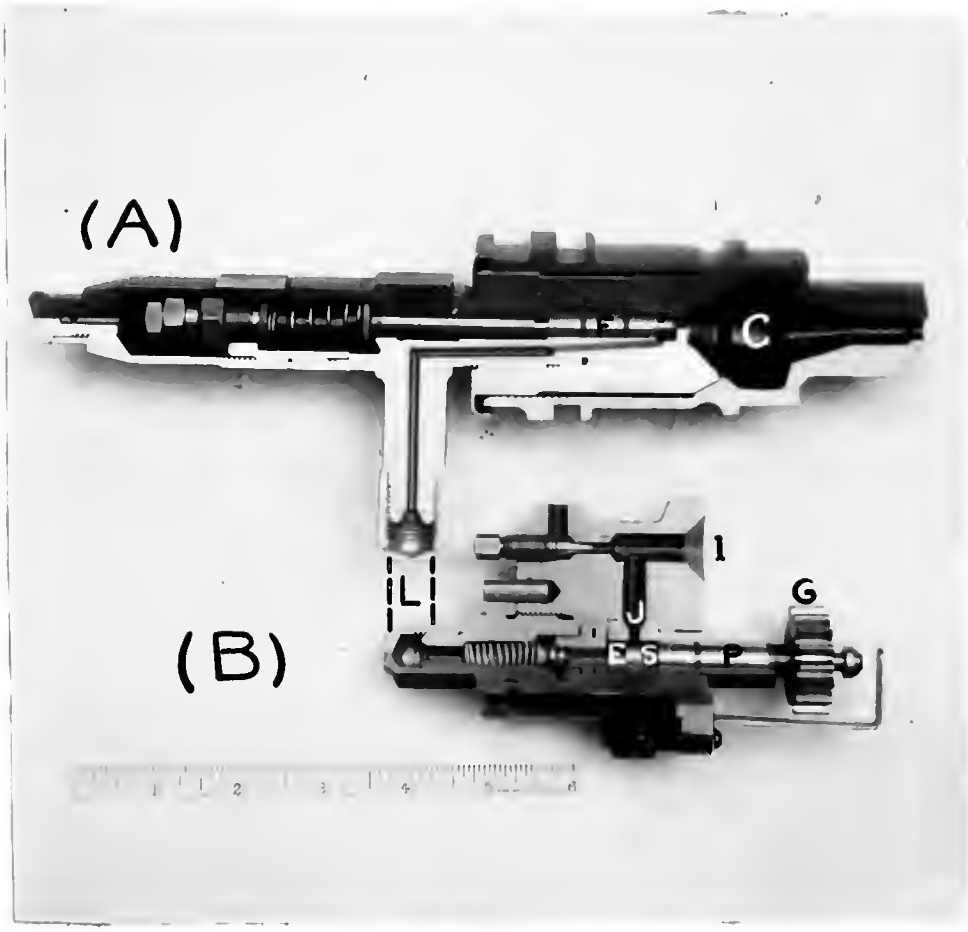


Figure 2.

[illegible]

[illegible] plunger face rotates [illegible] the left edge of the inlet channel "J". Further motion of [illegible] to the left causes the plunger to force oil from "S" past the check valve "V" until the left edge of the scroll space "S" reaches the right edge of the inlet channel "J". At this point the space "S" communicates directly with the inlet channel "J" through a small hole drilled from the plunger face to the scroll space "S", and the oil in space "S" is by-passed back to "J".

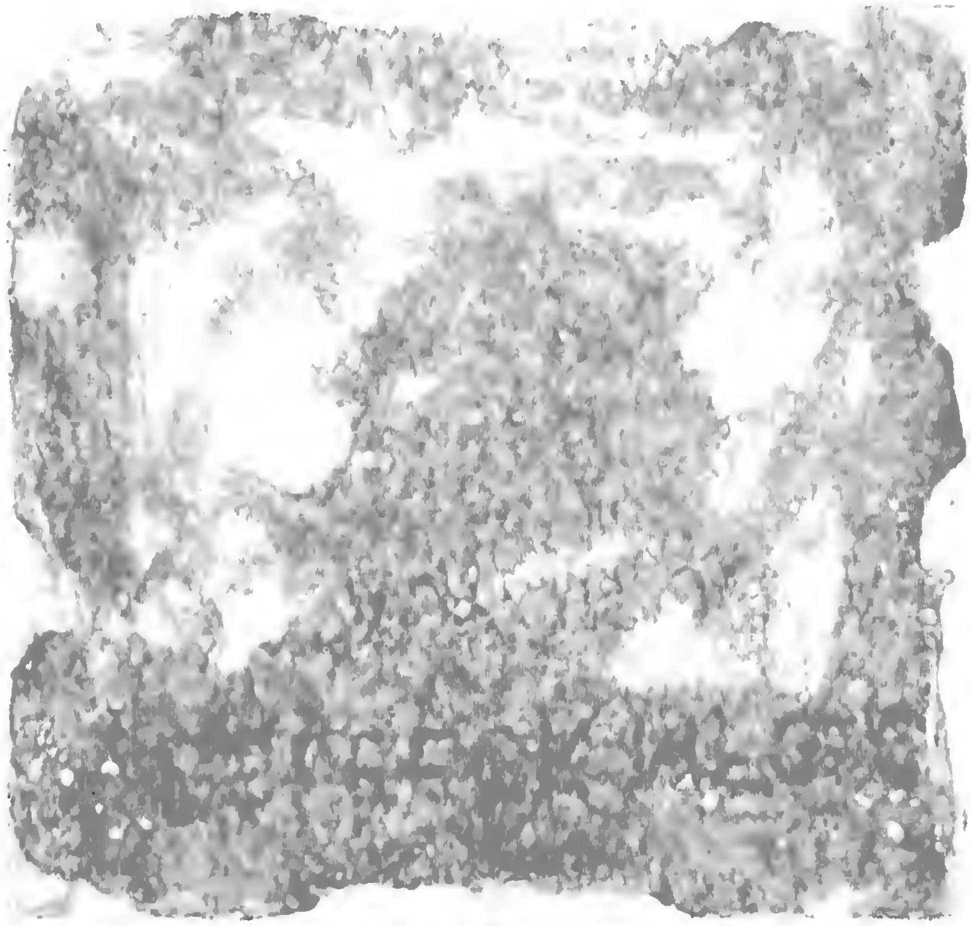


Figure 2.

Thus in this type of pump, delivery will always commence at the same angular shaft position, regardless of rack setting, whereas the angular shaft position at which oil ceases to be delivered will depend upon the setting of the rack.

Oil enters the pump Figure 2(B) at "I", is discharged past the check valve "V" from whence it is delivered to the nozzle valve, Figure 2(A) through the pipe line "L".

A Caterpillar Fuel Injection Valve designed for engines of $5\frac{1}{4}$" and 5-3/4" cylinder bore was used in this investigation. A picture of this valve, in cut-away section, is shown in Figure 2(A); the valve actually used, however, differs from the one shown in Figure 2(A) in that the pre-combustion chamber "C" was not used. The manufacturer's part designation for this valve is as listed below:

Part Name	Part Number	Weight (Grams)
Spray valve spring	1A6926	11.22
Spray valve spring stem	2A4684	16.07
Spray valve needle	2A4682	5.74

Other pertinent data pertaining to the nozzle valve are: (1) needle valve lift 0.007"; (2) needle valve stem diameter 0.039"; (3) included angle between faces of valve seat = 60°; (4) orifice length 0.118"; (5) orifice diameter 0.025"; (6) spring constant 771.2 lbs. per in. deflection; (7) clearances between "P" and "F" and their respective working surfaces = lap fit.

In obtaining the data necessary for plotting the curves shown

In Figures 2, 3 and 4, the adjustable spring of the injection valve was first carefully set with the aid of a hand pump to cause the valve to open at the desired pressure. The pump speed was maintained at a constant value and the number of pump revolutions required for the delivery of a known weight of oil, as indicated by the balance and indicator light system, was recorded on the revolution counter "R", Figures 1 and 4. Several "runs" were made for each rack setting to insure accuracy of data.

The spray penetration pictures shown in Figure 9 were taken at a pump speed of 600 R.P.M. and a rack setting of .400 (moderate load); those in Figure 10 were taken at the same pump speed but for a rack setting of .600 (light load); nozzle opening pressure in both cases was 1750 lbs. per sq.in. Exposures were taken for every one degree angular shaft displacement, starting with the first instant of injection and continuing well past the point of cut-off. Since a speed of 600 R.P.M. is equivalent to one revolution in a tenth of a second, the pictures were given a time exposure of one-tenth of a second, thereby giving only one injection per picture rather than a series of injections. The series of pictures represent several hundred injections rather than a time development of one injection.

The characteristic properties of the Diesel fuel used were as follows:

Gravity................... 30.8 A.P.I. at 60°F.

Viscosity .... 36.7 S.U.S. at 100°F.

Surface Tension .... 29.8 dyne/cm at 72°F.

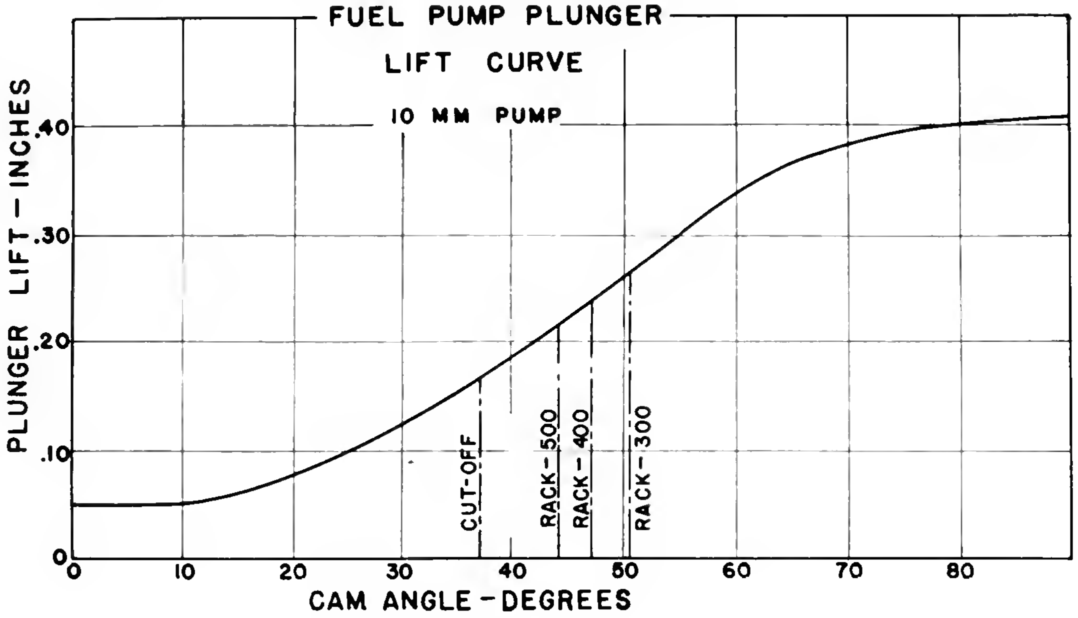


FIGURE 3

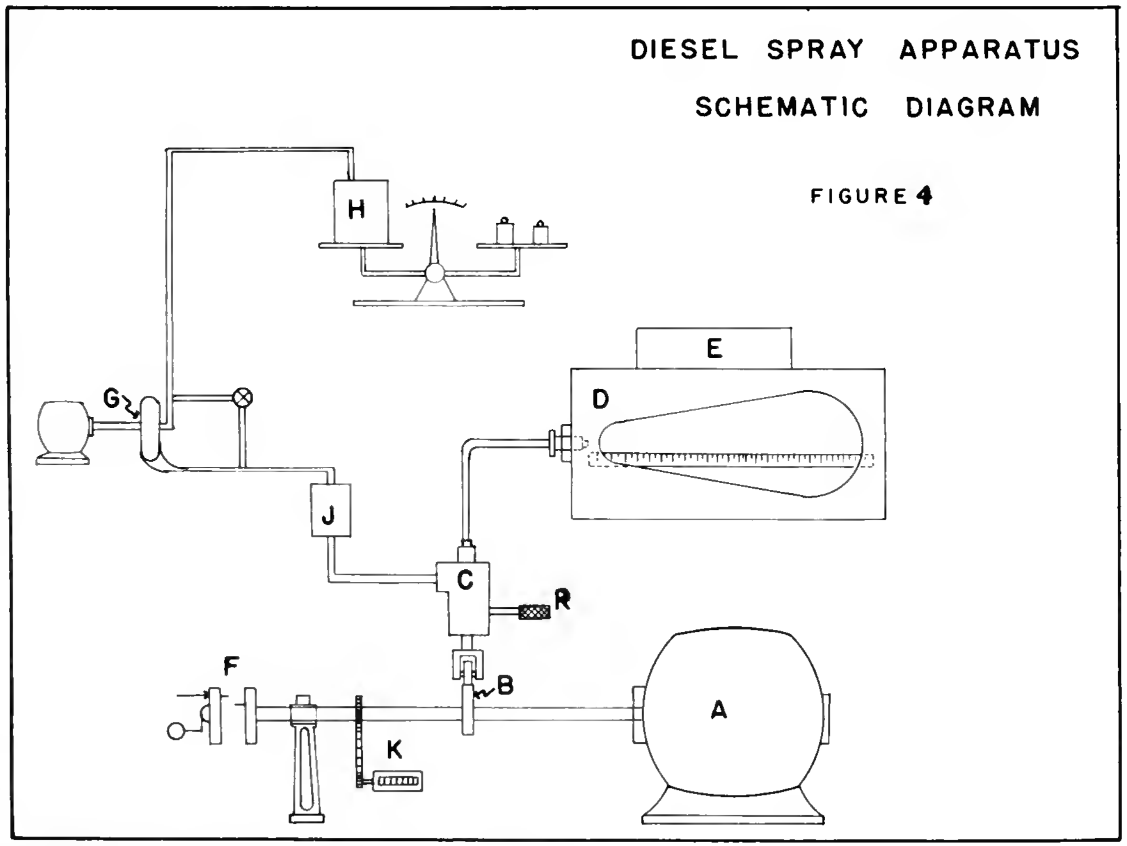


FIGURE 4

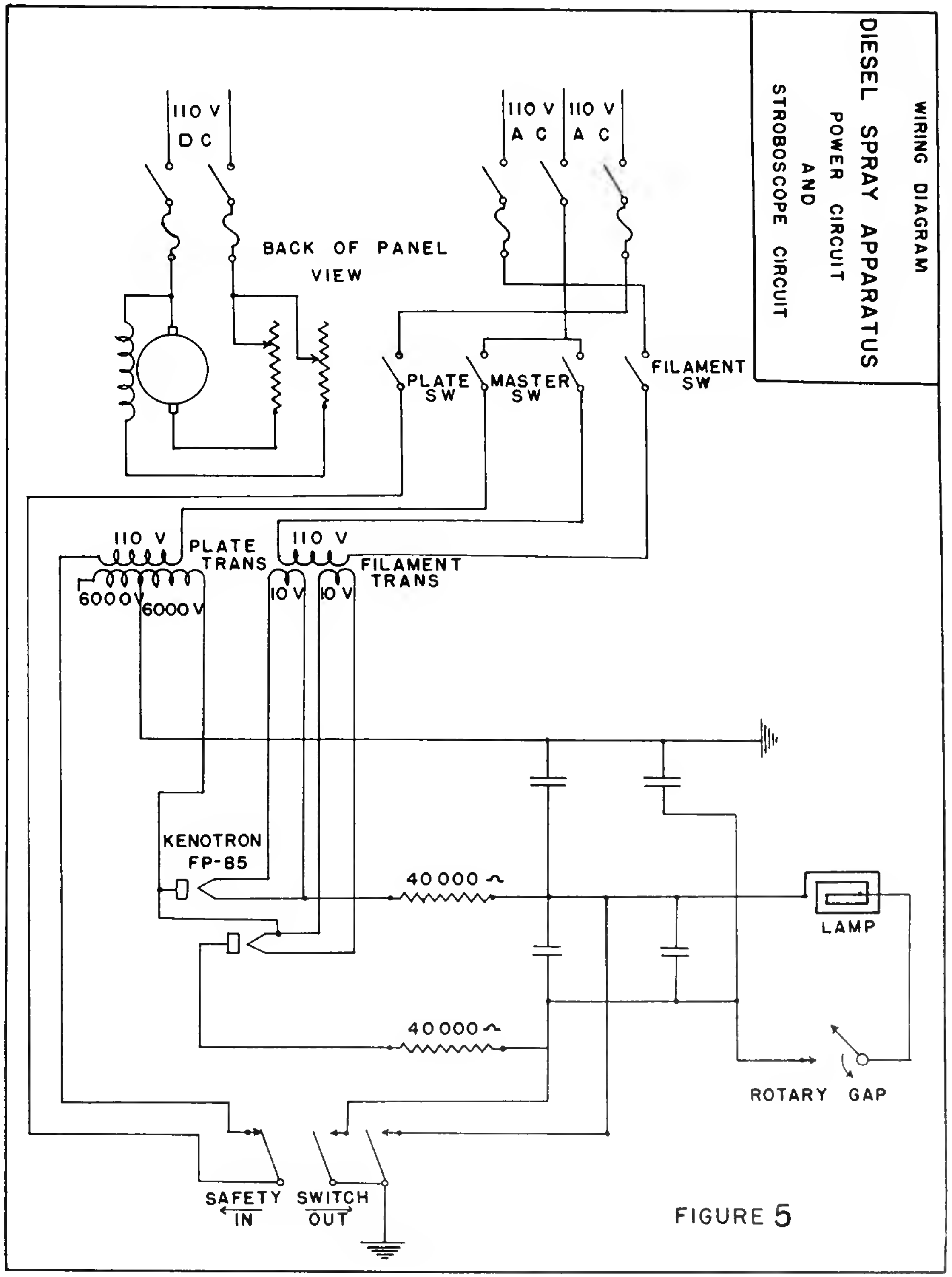


FIGURE 5

DISCUSSION AND RESULTS

The quantity of oil which the fuel pump, Figure 2(B), should deliver for various rack settings, based on displacement data, was computed as follows and is shown by the plane surfaces A B C D in Figure 6:

Dial indicator readings were carefully taken of the cam contour for the various angular crank positions from which the lift curve, Figure 3, was constructed. Then the angular shaft position for cut-off (the point at which the fuel pump, Figure 2(B), began delivery) was obtained by manually unseating the check valve and observing the point at which the oil ceased to flow from the fuel pump; the oil to the suction side of the pump being maintained under pressure due to the gravity head from supply tank "H", see Figure 1. This angular shaft position was, as should be expected, the same for all rack settings. Next, the angular position of the shaft at the point of release (that point at which the pump stopped delivery due to the scroll position), was determined for each rack position. This was accomplished by again lifting the fuel pump check valve manually, and observing the angular shaft position at which oil began to flow from the pump. These angular positions were then transferred to the lift curve, Figure 3, from which the effective pump stroke for any rack setting was obtained by subtracting the lift at the point of cut-off from the lift at the point of release. Knowing the effective pump stroke, for any rack setting, the pump plunger area and the density

[illegible]

The quantity of [illegible] delivered [illegible] for the four rack settings, [illegible], was computed as follows and is shown by the plane surfaces [illegible] in Figure [illegible]:

Dial indicator readings were carefully taken of the cam contour for the various angular crank positions from which the lift curve, Figure 3, was constructed. Then the angular shaft position for cut-off (the point at which the fuel pump, Figure 2(a), began delivery) was obtained by manually unseating the check valve and observing the point at which the oil ceased to flow from the fuel pump; the oil to the suction side of the pump being maintained under pressure due to the gravity head from supply tank "H", see Figure 1. This angular shaft position was, as should be expected, the same for all rack settings. Next, the angular position of the shaft at the point of release (that point at which the pump stopped delivery due to the scroll position), was determined for each rack position. This was accomplished by again lifting the fuel pump check valve manually, and observing the angular shaft position at which oil began to flow from the pump. These angular positions were then transferred to the lift curve, Figure 3, from which the effective pump stroke for any rack setting was obtained by subtracting the lift at the point of cut-off from the lift at the point of release. Knowing the effective pump stroke, for any rack setting, the pump plunger area and the density

of the Diesel oil, the weight of oil discharged per stroke was computed from the formula: weight per stroke = area of piston x stroke of piston x oil density. Since the area of the pump was .000846 square feet (plunger diameter 10 millimeters), and the density of the oil at 12 lbs. per sq. in. pressure and 74°F, was 62.284 x .8676 = 54.038 pounds per cubic foot, the weight of oil delivered per stroke was: $.000846 \times 54.038 \times \frac{\text{lift}''}{12} = .00381 \times \text{lift}''$ (lbs.). The results of the above computations are as tabulated:

TABLE I

Rack Setting	Cut Off	Release	Lift at Cut Off	Lift at Release	Effective Pump Stroke	Calculated Wt. of oil per Stroke (lbs.)
.150	37°	56°	.175"	.305"	.130"	4.96×10^{-4}
.175	"	55°	"	.297"	.122"	4.65×10^{-4}
.200	"	54°	"	.290"	.115"	4.38×10^{-4}
.225	"	53°	"	.282"	.107"	4.08×10^{-4}
.250	"	52°	"	.274"	.099"	3.77×10^{-4}
.300	"	50.5°	"	.262"	.087"	3.315×10^{-4}
.350	"	49.25°	"	.253"	.078"	2.97×10^{-4}
.400	"	47.25°	"	.238"	.063"	2.40×10^{-4}
.450	"	46°	"	.227"	.052"	1.98×10^{-4}
.500	"	44°	"	.213"	.038"	1.45×10^{-4}
.550	"	42.5°	"	.202"	.027"	1.03×10^{-4}
.600	"	40°	"	.185"	.010"	0.381×10^{-4}

From Figure 6, it may be observed that the surface representing the oil actually discharged per stroke for any given rack setting

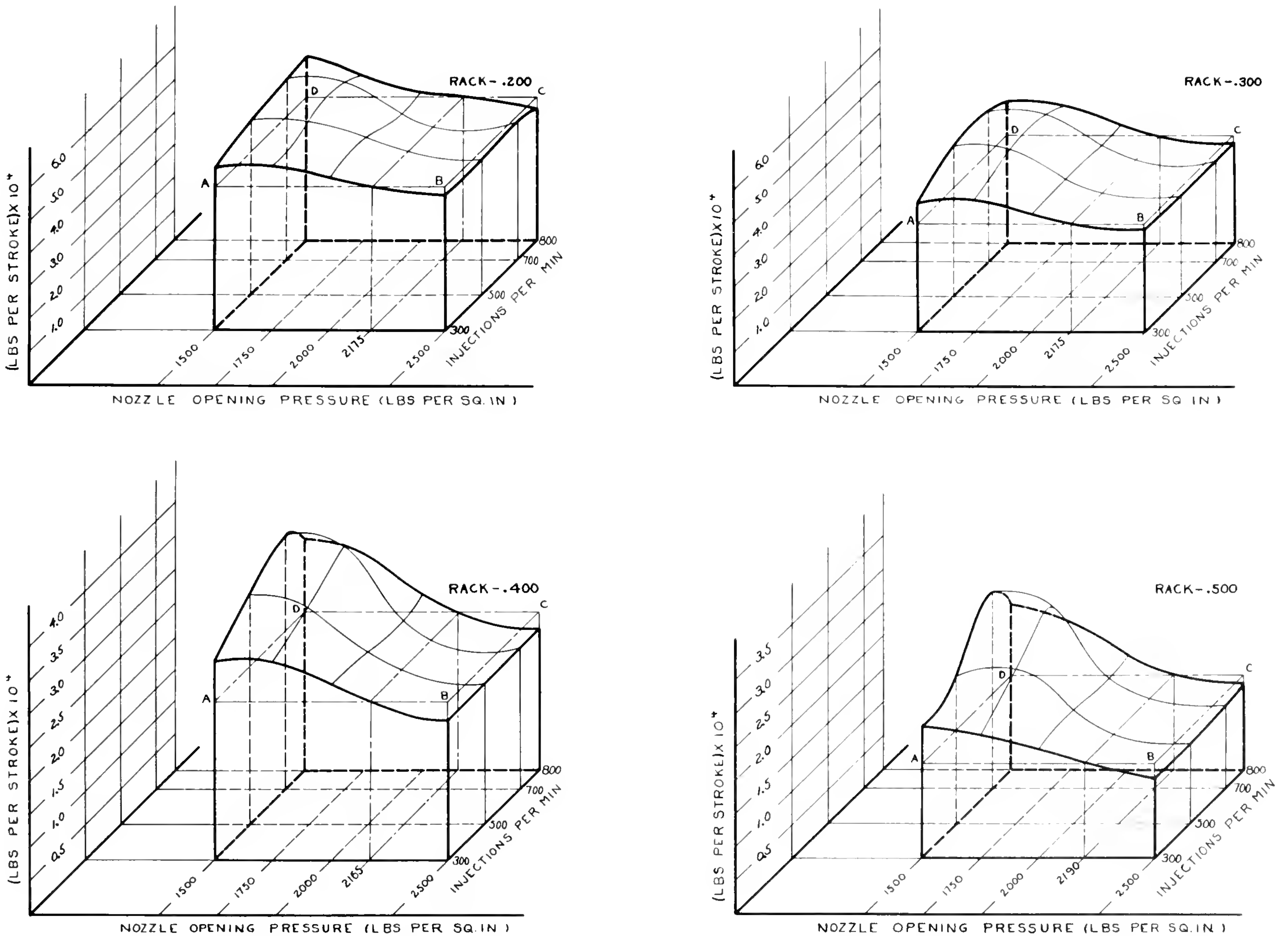
of the 'cool oil', the weight of oil discharged per stroke was computed from the [illegible] weight per stroke = area x [illegible] x stroke of piston x oil density. Since the size of the pump was [illegible] [illegible] (plunger diameter 10 millimeters), and the density of the oil at 12 lbs. per sq. in. pressure and 78°F, [illegible] 54.036 pounds per cubic foot, the weight of oil delivered per stroke was: .00840 x .022 x $\frac{.1176''}{12}$ = .0231 x .1176" (lbs.). The results of the above computations are as tabulated:

TABLE X

Rack Setting	Cut Off	Re-lease	Lift at Cut Off	Lift at Release	Effective Pump Stroke	Calculated Wt. of oil per Stroke (lbs.)
.150	27°	56°	.176"	.296"	.120"	4.96 x 10^{-4}
.175	"	55°	"	.297"	.122"	4.55 x 10^{-4}
.200	"	54°	"	.290"	.115"	4.38 x 10^{-4}
.225	"	53°	"	.282"	.107"	4.08 x 10^{-4}
.250	"	52°	"	.274"	.098"	3.77 x 10^{-4}
.300	"	50.5°	"	.262"	.087"	3.315 x 10^{-4}
.350	"	49.25°	"	.253"	.078"	2.97 x 10^{-4}
.400	"	47.25°	"	.238"	.063"	2.40 x 10^{-4}
.450	"	46°	"	.227"	.052"	1.98 x 10^{-4}
.500	"	44°	"	.213"	.038"	1.46 x 10^{-4}
.550	"	42.5°	"	.202"	.027"	1.03 x 10^{-4}
.600	"	40°	"	.185"	.010"	0.381 x 10^{-4}

From Figure 6, it may be observed that the surface representing the oil actually discharged per stroke for any given rack setting

THE INFLUENCE OF PUMP SPEED AND NOZZLE
OPENING PRESSURE
ON DISCHARGE RATE FOR VARIOUS RACK SETTINGS
PIPE LENGTH - 30 INCHES
FIGURE 6



changes contour with both speed and pressure variations. Disregarding the dynamics of the system this surface should be coincident with the plane surface ABCD which represents the amount of oil which the pump should deliver based on displacement data. Obviously the planes A B C D in Figure 6 are horizontal as shown since the theoretical weight delivered per stroke is independent of all variables save rack setting. That the weight per stroke contour surface does not coincide with the plane surfaces A B C D Figure 6, is clearly shown for the rack settings considered (.200, .300, .400, .500). The fact that the pump actually delivers more (or less) oil than is shown by displacement computations may be explained in the following way.(1) (2) (6)* Once the check valve has been lifted off its seat, due to the oil pressure created by the plunger motion, oil will flow into the line and pressure waves will develop between the face of the pump plunger and the nozzle valve. When the pump reaches the point of release, the check valve will remain off its seat for an appreciable interval due to its inertia as well as to the friction force occasioned by the viscous drag of the oil flowing past the valve. Thus the next pressure wave, after release, reflected from the plunger face will force additional oil past the open check valve and into the fuel line. This action is possible, in spite of the fact that the plunger by-pass channel puts the pump chamber in direct communication with the suction side of the pump, since the by-pass channel area is so small that the pressure wave moving toward the plunger face will be reflected from the plunger face before it has driven any oil out of the chamber space into the suction line. After the check valve has been seated, pressure

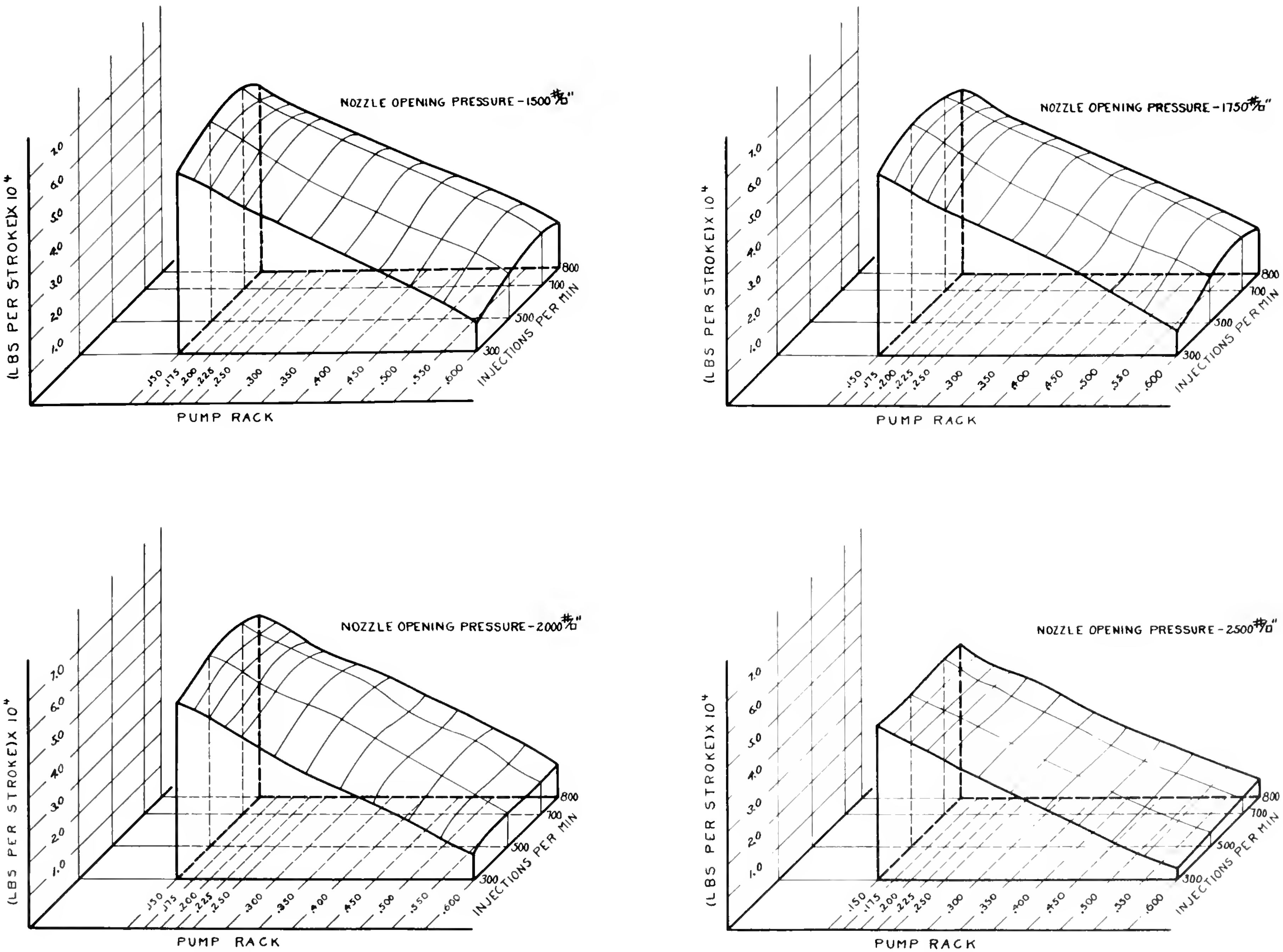
*Such designations refer to similar numbers in bibliography.

surges will exist between the face of the pump plunger and the check valve which may be of sufficient magnitude to reopen the check valve against the oil pressure in the line, thus forcing oil from the pump barrel into the oil line. The higher the oil pressure in the line, as determined by the nozzle valve opening pressure, the less will be the tendency for these pressure surges to re-open the check valve once it has become seated. This "super-charging" effect is clearly seen in Figure 6 for all rack settings and nozzle opening pressures below a certain value. The nozzle opening pressures at which "super-charging" does not exist are represented by the intersections of the plane surfaces A B C D, Figure 5, with their respective contour surfaces, and these values are:

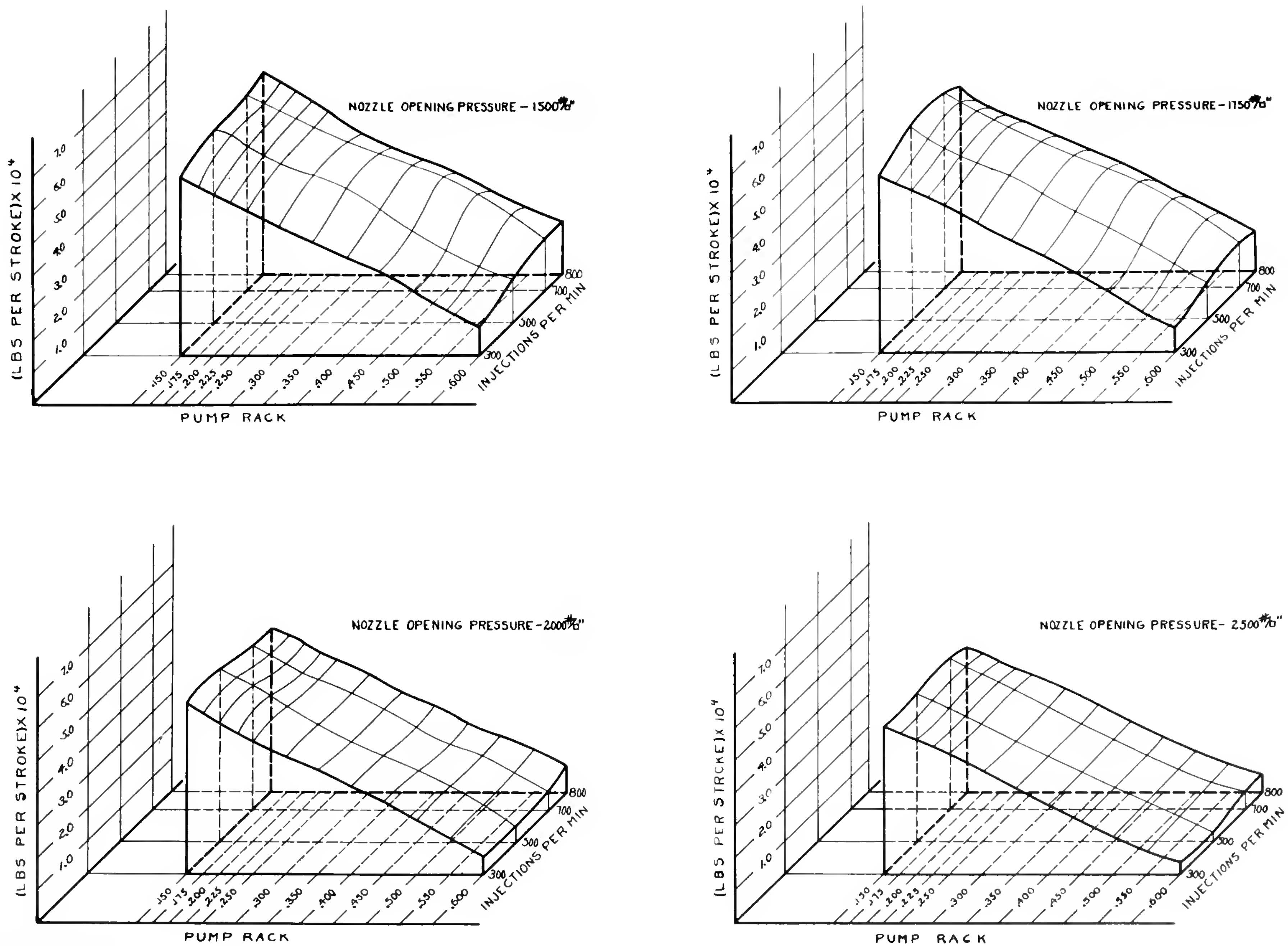
Rack setting	Nozzle opening pressure (lbs. per sq.in.)
.200	2175
.300	2175
.400	2165
.500	2190

It may be noted in Figure 6 that for the above nozzle opening pressures the actual quantity of oil delivered per stroke is independent of speed and varies linearly with rack setting. As the nozzle opening pressure was increased beyond the average value of 2175 lbs. per sq. in., the oil pressure in the line was high enough, after the point of release, to keep the pump check valve closed against the action of the pressure surges in the pump barrel. Under these conditions, the pump discharged less oil than is indicated by the plane

THE INFLUENCE OF PUMP SPEED AND RACK SETTING
ON DISCHARGE RATE
FOR VARIOUS NOZZLE OPENING PRESSURES
PIPE LENGTH - 15 INCHES
FIGURE 7



THE INFLUENCE OF PUMP SPEED AND RACK SETTING
ON DISCHARGE RATE
FOR VARIOUS NOZZLE OPENING PRESSURES
PIPE LENGTH - 30 INCHES
FIGURE 8



surfaces A B C D. The amount of oil actually delivered decreased with increasing nozzle opening pressures due to the greater force exerted by the oil to keep the check valve on its seat against the action of the pressure surges, and to the increased pump leakage occasioned by the higher discharge pressures.(5)

Figure 7 shows the influence of rack setting and speed on the weight of oil discharged per stroke for various nozzle opening pressures and a pipe length of 15 inches; Figure 8 shows a series of similar surfaces for a pipe length of 30 inches. Again it may be noted from these figures that as the nozzle opening pressure approaches 2175 lbs. per sq. in., the weight contour surface approaches plane surfaces; see Figures 7 and 8 for nozzle opening pressures of 2000 and 2500 lbs. per sq. in. These contour surfaces also clearly show the influence of pump speed changes on the quantity of oil discharged. Here it is to be noted that in general the weight of oil discharged increased with higher speed up to about 700 R.P.M.(3) while for the range between 700 -- 800 R.P.M., the weight of oil decreased;(5) this may be seen most clearly in Figures 7 and 8 and Table II for nozzle opening pressures of 1500 and 1750 lbs. per sq. in. For the nozzle opening pressures of 2000 and 2500 lbs. per sq. in. (in the neighborhood of 2175 lbs. per sq. in.) the influence of pump speed was not so marked.

Table II follows.

surfaces [illegible]. The amount of oil [illegible] with increasing nozzle opening pressure [illegible] exerted by the [illegible] action of the pressure surges, and [illegible] occasioned by the higher discharge pressures.[illegible]

Figure 7 shows the influence of rack setting and speed on the weight of oil discharged per stroke for various nozzle opening pressures and a pipe length of 15 inches. Figure 8 shows a series of similar surfaces for a pipe length of 30 inches. Again it may be noted from these figures that as the nozzle opening pressure approaches 2175 lbs. per sq. in., the weight contour surface approaches plane surfaces; see Figures 7 and 8 for nozzle opening pressures of 2000 and 2500 lbs. per sq. in. These contour surfaces also clearly show the influence of pump speed changes on the quantity of oil discharged. Here it is to be noted that in general the weight of oil discharged increased with higher speed up to about 700 R.P.M.,[5] while for the range between 700 -- 800 R.P.M., the weight of oil decreased;[5] this may be seen most clearly in Figures 7 and 8 and Table II for nozzle opening pressures of 1500 and 1750 lbs. per sq. in. For the nozzle opening pressures of 2000 and 2500 lbs. per sq. in. (in the neighborhood of 2175 lbs. per sq. in.) the influence of pump speed was not so marked.

Table II follows.

TABLE II

Rack Setting	Pump Speed	ACTUAL DISCHARGE (LBS.PER STROKE X 10^6);Pipe Length 30". Nozzle Opening Pressure (Lbs. Per Sq. In.) 1500	1750	2000	2500
.200	300	498	505	467	418
	500	541	535	485	412
	700	556	550	450	424
	800	568	508	457	403
.300	300	394	404	366	318
	500	459	452	365	311
	700	460	457	359	312
	800	435	422	361	319
.400	300	303	304	269	216
	500	352	337	265	216
	700	388	373	257	217
	800	355	336	270	217
.500	300	202	191	166	122
	500	226	236	173	123
	700	300	279	167	128
	800	254	228	174	132

The fact that the quantity of oil increased for increased pump speed up to 700 R.P.M., and then decreased for further speed increases may be explained as follows: higher pump speeds give higher plunger speed, and hence impart greater velocities to the oil being discharged from the pump; this increase in kinetic energy causes a greater quantity of oil to flow past the pump check valve after release and before the check valve has become seated. As the pump speed increased beyond a certain value, however, the volumetric efficiency of the pump decreased since oil could not flow into the pump chamber fast enough to completely fill it, and the quantity of oil discharged decreased.

A comparison of Figures 7 and 8 shows pipe length, for the two

[illegible]

Rack setting	Pump speed	[illegible]	[illegible]	[illegible]	[illegible]
		[illegible]	[illegible]	1000	[illegible]
.200	300	[illegible]	[illegible]	[illegible]	[illegible]
	500	[illegible]	[illegible]	[illegible]	[illegible]
	700	[illegible]	[illegible]	[illegible]	[illegible]
	800	[illegible]	[illegible]	[illegible]	[illegible]
.300	300	704	[illegible]	[illegible]	[illegible]
	500	452	[illegible]	300	[illegible]
	700	[illegible]	[illegible]	[illegible]	[illegible]
	800	[illegible]	[illegible]	261	[illegible]
.400	300	[illegible]	[illegible]	[illegible]	216
	500	[illegible]	[illegible]	265	[illegible]
	700	388	[illegible]	267	217
	800	355	[illegible]	270	217
.500	300	202	[illegible]	166	122
	500	226	[illegible]	173	128
	700	300	[illegible]	167	128
	800	254	228	174	132

The fact that the quantity of oil increased for increased pump speed up to 700 R.P.M., and then decreased for further speed increases may be explained as follows: higher pump speeds give higher plunger speed, and hence impart greater velocities to the oil being discharged from the pump; this increase in kinetic energy causes a greater quantity of oil to flow past the pump check valve after release and before the check valve has become seated. As the pump speed increased beyond a certain value, however, the volumetric efficiency of the pump decreased since oil could not flow into the pump chamber fast enough to completely fill it, and the quantity of oil discharged decreased.

A comparison of Figures 7 and 8 shows pipe length, for the two

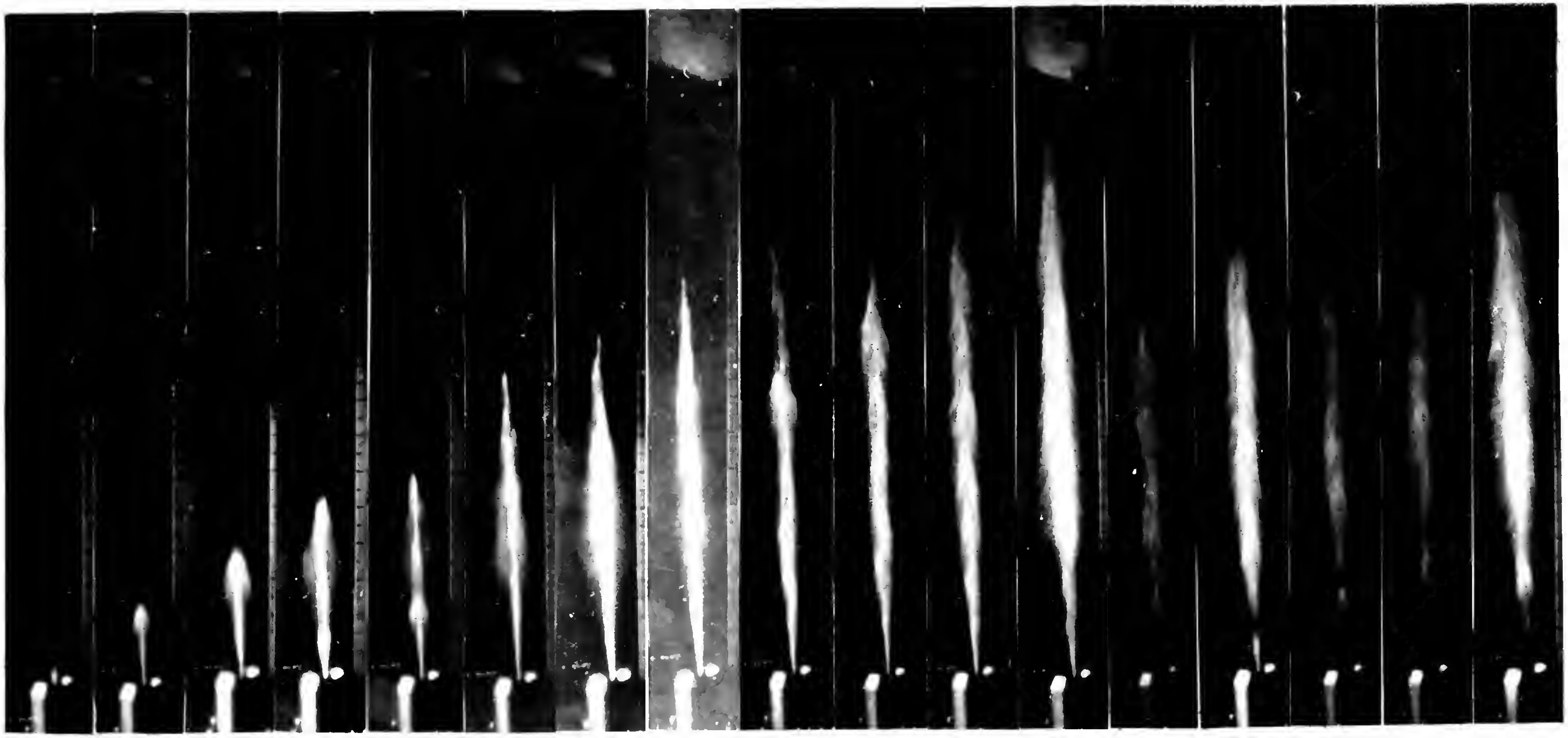


Figure 9

SPRAY DEVELOPMENT FOR AVERAGE OF SEVERAL HUNDRED INJECTIONS.

Time Between Frames, .000278 seconds
Nozzle Opening Pressure, 1750 lbs. per sq. in.
Pump Speed, 600 R.P.M.
Length of Pipe Line, 15 inches
Rack Setting, .400 (2.40 x 10^4 lbs. of oil per stroke)
Chamber Pressure, Atmospheric

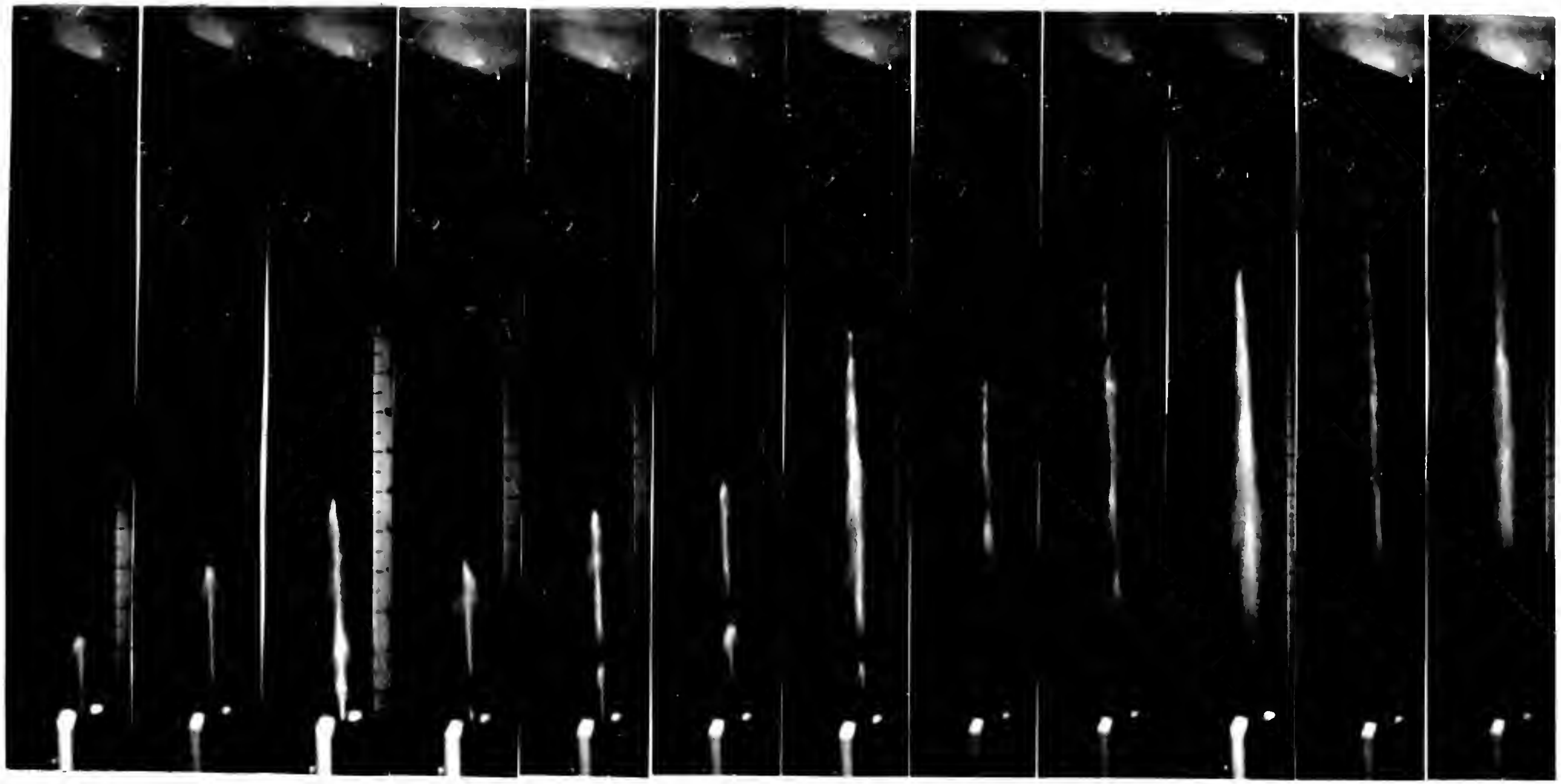


Figure 10

SPRAY DEVELOPMENT FOR AVERAGE OF SEVERAL HUNDRED INJECTIONS.

Time Between Frames, .000278 seconds
Nozzle Opening Pressure, 1750 lbs. per sq. in.
Pump Speed, 600 R.P.M.
Length of Pipe Line, 15 inches
Rack Setting, .600 (.381 x 10^4 lbs. of oil per stroke)
Chamber Pressure, Atmospheric.

lengths investigated, to have little influence.[3] For nozzle opening pressures of 1750 and 2500 lbs. per sq. in., the two pipe lengths gave almost identical contour surfaces, while for nozzle opening pressures of 1500 and 2000 lbs. per sq. in. some irregularities between the contour surfaces may be noted. These irregularities, as may be seen, occurring for pump speeds of 500 R.P.M. and above.

A comparison of Figures 9 and 10 show the influence of rack setting on penetration at a pump speed of 600 R.P.M., and a nozzle opening pressure of 1750 lbs. per sq. in., when discharging against atmospheric pressure. Figure 9 is for a rack setting of .400, while Figure 10 is for a rack setting of .600. For both rack settings, the pictures marked 1 were taken at two degrees of pump shaft angular displacement after the point of injection was observed which is equivalent to one eighteen-hundredth of a second. Thereafter the pictures were taken in succession at one degree pump shaft intervals (one thirty-six-hundredth of a second). In Figure 9, evidence of secondary discharges may be seen in pictures 4, 5, 13, 14, 15 and 17.[4] Maximum penetration of 16.5 inches is shown in picture 12 and cut off in picture 13. The depth of penetration is quite uniform up to the point of cut off, picture 13. It is interesting to note at this point that Figure 9 shows a definite injection period over an interval of 13 degrees angular pump shaft displacement, whereas the table on page 12 for a rack setting of .400 shows the injection period to be only 10.25 degrees. This point again established the fact that the pump actually discharges more oil under certain conditions than is theoretically possible.

Lengths [illegible] ing pressures [illegible] gave [illegible] since [illegible] the [illegible] surfaces may [illegible] occur, according for [illegible].

A comparison of [illegible] setting on penetration at a pump speed of [illegible] r.p.m., and nozzle opening pressure of 1700 lbs. per sq. in., when discharging against atmospheric pressure. Figure 8 is for a rack setting of .400, while Figure 10 is for a rack setting of .300. For both rack settings, the pictures marked 1 were taken at two degrees of pump shaft angular displacement after the point of injection was observed which is equivalent to one eighteen-hundredth of a second. Thereafter the pictures were taken in succession at one degree pump shaft intervals (one thirty-six-hundredth of a second). In Figure 8, evidence of secondary discharges may be seen in pictures 4, 5, 13, 14, 15 and 17.(4) Maximum penetration of 15.5 inches is shown in picture 12 and cut off in picture 13. The depth of penetration is quite uniform up to the point of cut off, picture 13. It is interesting to note at this point that Figure 8 shows a definite injection period over an interval of 13 degrees angular pump shaft displacement, whereas the table on page 12 for a rack setting of .400 shows the injection period to be only 10.25 degrees. This point again established the fact that the pump actually discharges more oil under certain conditions than is theoretically possible.

In Figure 10 it may be readily seen that the spray penetration is quite irregular. Secondary discharges may be noted in pictures 3, 4, 5, 6, 7, 9 and 10. Cut off has taken place in picture 7, and here the maximum penetration was about 10 inches. Again it may be noted in Figure 10 that injection took place over a pump shaft interval of 8 degrees whereas the table on page 12, for a rack setting of .600, shows this interval to be only 3 degrees.

CONCLUSIONS

1. From Figure 6 it may be concluded that for this system there exists a certain nozzle opening pressure for which a linear relation exists between weight of fuel discharged per stroke and rack setting, and further that this linear relation is independent of speed for the range investigated; that is to say, for a pump speed from 300 -- 800 R.P.M. and a nozzle opening pressure of from 1500 -- 2500 pounds per sq. in. In this case the nozzle opening pressure required to give this linear relation was found to be about 2175 pounds per sq. in.

It appears reasonable to suppose that a similar nozzle opening pressure will exist, and may be found, for any injection system of the type investigated.

2. A comparison of Figures 7 and 8 shows that for the pipe lengths investigated there was little or no change in the weight of oil delivered per stroke for the two pipe lengths.

3. Figures 6, 7 and 8 show definitely that the quantity of oil discharged per stroke decreases with increase in the nozzle opening pressure.

4. Figures 6, 7 and 8 show that when operating at a nozzle opening pressure other than 2175 pounds per sq. in., as mentioned in 1 above, the weight of oil discharged per stroke will vary with pump speed, and the greater the departure of the pressure

CONCLUSIONS

1. [illegible] figures [illegible] that for this system there exists a certain [illegible] opening pressure [illegible] rack setting, and further that [illegible] of speed for this [illegible] speed from [illegible] 1600 -- 2400 pounds per sq. in. [illegible] pressure required to give this linear relationship [illegible] about 2175 pounds per sq. in.

It appears reasonable to suppose that a definite nozzle opening pressure will exist, at least within limits, for any injection system of the type investigated.

2. A comparison of figures 7 and 8 shows that for the pipe lengths investigated there was little or no change in the weight of oil delivered per stroke for the two pipe lengths.

3. Figures 6, 7 and 8 show definitely that the quantity of oil discharged per stroke decreases with increase in the nozzle opening pressure.

4. Figures 6, 7 and 8 show that when operating at a nozzle opening pressure other than 2175 pounds per sq. in., as mentioned in 1 above, the weight of oil discharged per stroke will vary with pump speed, and the greater the departure of the pressure

from this value of nozzle opening pressure, the greater will become the influence of pump speed. Generally speaking, the quantity of oil delivered per stroke, for the system investigated, increased with higher pump speed up to about 700 R.P.M., and further speed increase caused the quantity of oil discharged to decrease.

5. The quantity of oil delivered per stroke follows in a general way the rack setting, as seen from Figures 7 and 8, but this relation is not linear unless the nozzle is set to open at the proper spring setting (2175 pounds per sq. in. for this system); see Figure 6.

6. For a moderate load (rack setting of .400), it may be noted from Figure 9 that the spray penetration is reasonably uniform and that there is little evidence of secondary discharges. However, for light loads (rack setting of .600), it may be seen from Figure 10 that the spray penetration is irregular and that there is much evidence of secondary discharges.

ACKNOWLEDGMENT

The author desires especially to thank Professor C. J. Vogt of the Department of Mechanical Engineering, University of California, who started this investigation and who has given untiringly and cheerfully of both his time and wisdom in order that this problem might be carried forward.

Thanks are also extended to Lieutenant H. S. Persons, U.S.N., who assisted in obtaining all the necessary data and the taking and developing of the pictures contained herein; and to Mr. J. E. Gullberg, Lecturer in the Department of Zoology, University of California, for the assistance given in the taking of many pictures.

Finally, I desire to thank members of the W. P. A. Project in the Department of Mechanical Engineering for their work in constructing the experimental station and making necessary alterations.

ACKNOWLEDGMENT

The author desires especially to thank Professor [illegible] of the Department of Mechanical Engineering, University of California, who started this research [illegible] and who has given unstintingly and cheerfully of both his time and advice in order that this problem might be carried forward.

Thanks are also extended to Lieutenant [illegible], [illegible], who assisted in obtaining all the necessary data and the taking and developing of the pictures contained herein; and to Mr. [illegible] Gullberg, Lecturer in the Department of Zoology, University of California, for the assistance given in the taking of many pictures.

Finally, I desire to thank members of the W. P. A. Project in the Department of Mechanical Engineering for their work in constructing the experimental station and making necessary alterations.

BIBLIOGRAPHY

(1) Fuel Injection Mechanisms and Vibrations in Fuel Lines from "Solid Injection Diesels" by F. Sass.

(2) N. A. C. A. Technical Report No. 396.

(3) N. A. C. A. Technical Report No. 433.

(4) "Diesel Notes" by Carl J. Vogt.

(5) The Automobile Engineer, May 1934. "Jerk-Pump Injection" by K. J. DeJuhasz and P. G. Burman.

(6) A. S. M. E. Transactions for February 1938. "Determination of the Rate of Discharge in Jerk-Pump Fuel-Injection Systems", by K. J. DeJuhasz.

Thesis- 6315
M815 Moore
An investigation of some of the characteristics of a jerk pump injection system for diesel engines.